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Signed this October 23, 2006

Mitsuru NAKANO

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[Title of the Invention] INFINITE VARIABLE-SPEED DRIVE

[Claims]

[Claim 1] An infinite speed-variable drive which is coupled to a drive source and which has an input shaft rotatively driven by said drive source, an output shaft for extracting power derived from rotation of said input shaft, and a toroidal infinite speed-variable transmission and a planetary gear transmission disposed between said input shaft and said output shaft, wherein said toroidal infinite speed-variable transmission comprises a pair of input disks, each rotating in association with rotation of said input shaft; an output disk which is disposed concentrically with said respective input disks around an intermediate section of said input shaft while a side surface thereof is oriented opposite said inner side surfaces of said respective input disks, and which is rotatable independently of said input disks; plural trunnions which are associated with each pair of said input disk and said output disk; support shafts each associated with each of the trunnions; and power rollers each associated with each of the trunnions and each sandwiched between side surfaces of respective pair of the input disk and the output disk; wherein said planetary gear mechanism is composed by engaging a planetary gear with a sun gear and a ring gear, the planetary gear disposed between the sun gear and the ring gear which is disposed around the

sun gear, the planetary gear rotatively supported by said carrier which is supported concentrically with said sun gear and supported on one end section of said input shaft; wherein transmission of power between said carrier and said one input disk is made free by means of causing a plurality of projection sections to engage with a plurality of engagement sections, the plurality of projection sections projectingly provided on portions of an outer side surface of said one input disk, said portions corresponding to areas close to the outermost diameter of an inside surface of said input disk where a traction section serving as a roll contact section between said inner side surface of said input disk and a circumferential surface of said power roller is located, the plurality of engagement sections formed in a part of said carrier.

[Claim 2] The infinite speed-variable drive according to claim 1, wherein said plurality of projection sections are projectingly provided at positions closer to the radially outside than to a circumscribed circle of a contact oval existing in said traction section.

[Claim 3] The infinite speed-variable drive according to claim 1, wherein there can be achieved the state of an infinite gear ratio in which said output shaft is stopped while said input shaft is being rotated.

[Detailed Description of the Invention]

[0001]

[Technical Field to which the Invention Pertains]

An infinite variable-speed drive according to the present invention is utilized as an automatic transmission for use with an automobile.

[0002]

[Related Art]

A toroidal infinite variable-speed drive has already been known as a kind of transmission constituting a transmission for use with an automobile. Such a toroidal infinite variable-speed drive that has already been put into practice in some applications in the aforementioned manner is of so-called double cavity type, wherein power is transmitted from an input section to an output section by way of two separate systems provided in parallel with each other. Toroidal infinite variable-speed drives described in patent documents 1 and 2 and those described in numerous other publications have already been known as examples of such a toroidal infinite variable-speed drive. A basic structure of the toroidal infinite variable-speed drive will be described by reference to Fig. 8.

[0003]

The toroidal infinite variable-speed drive shown in Fig. 8 has an input shaft 1. Input disks 2, 2 are provided at respective areas close to an input base end section (i.e., a left-side portion in Fig. 8) and an extremity (i.e., a right-side

portion in Fig. 8) of the input shaft 1. The input disks 2, 2 support ball splines 4, 4 such that input side surfaces 3, 3 consisting of toroidal surfaces are mutually opposed with respect to the input shaft 1. The input disks 2, 2 are supported so as to rotate in synchronism with the input shaft 1 such that the input disks 2, 2 can be displaced in the axial direction of the input shaft 1.

[0004]

A rolling bearing 5 and a pressing apparatus 6 of coding cam type are interposed between a base end section of the input shaft 1 (the left end section shown in Fig. 8) and an exterior surface of the input disk 2. A cam plate 7 constituting the pressing apparatus 6 is set so that the cam plate can be rotatably driven by a drive shaft 8. A loading nut 9 and a flat leaf spring 10 having great resiliency are interposed between the extremity (the right end shown in Fig. 8) of the input shaft 1 and an outer side surface of the input disk 2.

[0005]

An intermediate section of the input shaft 1 penetrates through a through hole 13 formed in a partition section 12 provided within a casing 11 housing the toroidal infinite variable-speed drive (see Fig. 1 showing an embodiment of the present invention). A cylindrical output cylinder 14 is rotatably supported by a pair of roll bearings 15, 15 on an internal diameter side of the through hole 13. An output gear

16 is fixedly fitted around an outer periphery of the intermediate section of the output cylinder 14. A pair of output disks 17, 17 are provided on the respective ends of the output cylinder 14 which protrude from outside surfaces of the partition section 12 and are supported so as to rotate in synchronism with the output cylinder 14 by means of spline engagement. A structure for rotatively coupling the pair of output disks to both ends of the output cylinder and a structure for coupling inside-diameter portions of the outside surfaces of the output disks to end sections of a sleeve by means of irregularity engagement, as described in Patent Document 3, have already been known as structures for rotatively coupling the pair of output disks to the respective end sections of the output cylinder.

[0006]

In this state, output side surfaces 18, 18 of the output disks 17, 17, the side surfaces consisting of toroidal surfaces, oppose the input side surfaces 3, 3. Alternatively, needle bearings 19, 19 are interposed between some portions of inner peripheral surfaces of the output disks 17, 17 protruding from the edge of the output cylinder 14 and an outer peripheral surface of the intermediate section of the input shaft 1. Rotation and axial fluctuations of the output disks 17, 17 with respect to the input shaft 1 are made variable while the load exerted on the output disks 17, 17 is supported.

[0007]

A plurality of power rollers 20, 20 (in general, two or three) are provided in a space (cavity) defined between the input and output side surfaces 3 and 18 around the input shaft 1. Each of the power rollers 20, 20 has a circumferential surface 21 which is brought into contact with the input and output side surfaces 3, 18 and formed into a spherical protruding surface. The power roller 20 is supported on an inner side of a trunnion 22 by means of a support shaft 23 and a plurality of roller bearings so as to be rotatable and slightly swayable.

[0008]

The trunnion 22 supports an axle provided on both ends thereof (i.e., with respect to a direction from the front side to the rear side of Fig. 8) on a support plate 24 (see Figs. 1 through 3 showing an embodiment of the present invention) provided within the casing 11 such that the axle is swayable and displaceable in the axial direction. The trunnion 22 supports the axle such that the axle becomes freely movable in a counterclockwise direction and a clockwise direction in Fig. 8 and is displaced in the axial direction of the axle (i.e., the vertical direction in Fig. 1 and the direction from the front side to the rear side of Fig. 8) by means of an unillustrated actuator.

[0009]

When the toroidal infinite variable-speed drive having

the foregoing construction is driven, the input disk 2 is rotatively driven by way of the pressing apparatus 6 by means of the drive shaft 8. The pressing apparatus 6 rotatively drives the input disk 2 while generating axial thrust. The pair of input disks 2, 2 including that input disk 2 are rotated synchronously while being pressed against the respective output disks 17, 17. Consequently, rotation of the input disks 2, 2 is transmitted to the respective output disks 17, 17 by way of the respective power rollers 20, 20. The output gear 16 coupled to the respective disks 17, 17 is rotated by way of the output cylinder 14.

[0010]

When a gear ratio between the drive shaft 8 and the output gear 16 is changed, the trunnions 22, 22 are displaced in the direction from the front side to the rear side of Fig. 8 by means of an unillustrated actuator. In this case, the trunnions 22, 22 of an upper half section in Fig. 8 and the trunnions 22, 22 in a lower half section in Fig. 8 are displaced in opposite directions to the same extent. The force applied, in a tangential direction, to a contact section between the circumferential surface 21 of the power roller 20 and the output side surfaces 3, 18 is changed in association with such a displacement. By means of the force in the tangential direction, the trunnions 22, 22 are swayed around the axles provided on the respective end sections thereof.

[0011]

In association with the swaying action, the positions of the contact sections between the circumferential surface 21 of the power roller 20 and the input and output side surfaces 13, 18 are changed with respect to the radial directions of the side surfaces 3, 18. As the contact sections change toward the outside with respect to the radial direction of the input side surface 3 and the inside with respect to the radial direction of the output side surface 18, the gear ratio is changed to acceleration. As the contact section changes toward the inside with respect to the radial direction of the input side surface 3 and the outside with respect to the radial direction of the output side surface 18, the gear ratio is changed to deceleration.

[0012]

For the case where the toroidal infinite variable-speed drive that is constructed and operates in the aforementioned manner is built in an actual automobile, construction of an infinite variable-speed drive by combination of the variable-speed drive with a differential unit of gear type, such as a planetary gear mechanism, has already been proposed. Fig. 9 shows an infinite variable-speed drive described in Pantent Document 4 from among the infinite variable-speed drives which have already been proposed. This infinite variable-speed drive is of a so-called geared neutral type, wherein rotation

of an output shaft can be switched between forward rotation and rearward rotation with a stop therebetween while the input shaft is being rotated in one direction. The infinite speed-variable drive is constructed by combination of a toroidal speed-variable transmission 25 with a planetary gear transmission 26. Of these transmissions, the toroidal infinite speed-variable transmission comprises the input shaft 1; the pair of input disks 2, 2; an output disk 17a; and the plurality of power rollers 20, 20. In the illustrated example, the output disk 17a has a structure in which a pair of output disks are integrated together by means of abutting the outer surfaces of the disks against each other.

[0013]

The planetary gear transmission 26 comprises the input shaft 1, and a carrier 27 fixedly coupled to one input disk 2 (i.e., the right input disk shown in Fig. 9). A first transmission shaft 29 having planetary gears 28a, 28b secured on respective ends of the shaft is rotatively supported by an intermediate portion of the carrier 27 in the radial direction thereof. A second transmission shaft 31 having sun gears 30a, 30b secured on respective ends thereof is rotatively supported in a manner concentric with the input shaft 1 on the side opposite the input shaft 1 with the carrier 27 sandwiched therebetween. The planetary gears 28a, 28b are engaged with a sun gear 33 fixed to the extremity (i.e., the right end section in Fig.

9) of a hollow rotary shaft 32 whose base end section (i.e., the left end section in Fig. 9) is coupled to the output disk 17a, or with the sun gear 30a fixed to one end section (i.e. the left end section in Fig. 9) of the second transmission shaft 31. One of the planetary gears 28 (i.e., the left planetary gear 28a shown in Fig. 9) is meshed with a ring gear 35 rotatively provided around the carrier 27 by way of the other planetary gear 34.

[0014]

Planetary gears 37a, 37b are rotatively supported on a second carrier 36 provided around the sun gear 30b fixed to the other end section (i.e., the right end section in Fig. 9) of the second transmission shaft 31. The second carrier 36 is disposed concentrically with the input shaft 1 and the second transmission shaft 31 and fixed to the base end section (i.e., the left end section in Fig. 9) of the output shaft 38. The planetary gears 37a, 37b mesh with each other, and the planetary gear 37a is engaged with the sun gear 30b, and the other planetary gear 37b is engaged with a second ring gear 39 rotatively provided around the second carrier 36. The ring gear 35 and the second carrier 36 are removably engaged with a low-speed clutch 40, and the second ring gear 39 and a stationary portion of a housing are removably engaged with a high-speed clutch 41.

[0015]

In the case of the infinite speed-variable drive shown

in Fig. 9 such as that described previously, the power of the input shaft 1 is transmitted to the output shaft 38 by way of the ring gear 35 in a so-called low-speed mode in which the low-speed clutch 40 is connected and the high-speed clutch 41 is disconnected. The gear ratio of the entire infinite speed-variable drive; that is, a gear ratio of the input shaft 1 to the output shaft 38, is changed by means of changing the gear ratio of the toroidal infinite speed-variable gear 25. In such a low-speed mode, the gear ratio of the entire infinite speed-variable drive is changed infinitely. Specifically, the rotation of the output shaft 38 can be switched between forward and rearward with a stop therebetween while the input shaft 1 is rotated in one direction, by means of adjusting the gear ratio of the toroidal infinite speed-variable transmission 25.

[0016]

During acceleration or constant-speed traveling of the automobile in such a low-speed mode, the torque that has passed through the toroidal infinite speed-variable drive 25 (i.e., passing torque) is applied to the output disk 17a from the input shaft 1 by way of the carrier 27, the first transmission shaft 29, the sun gear 33, and the hollow rotary shaft 32. The torque is further applied from the output disk 17a to the input disks 2, 2 by way of the power rollers 20, 20. Specifically, the torque passing through the toroidal infinite speed-variable drive 25 during acceleration or constant-speed driving is

circulated in the direction in which the input disks 2, 2 experience the torque output from the power rollers 20, 20.

[0017]

In a so-called high-speed mode in which the low-speed clutch 40 is disconnected and the high-speed clutch 41 is connected, the power of the input shaft 1 is transmitted to the output shaft 38 by way of the first and second transmission shafts 29, 31. The gear ratio of the entire infinite speed-variable drive is changed by means of changing the gear ratio of the toroidal infinite speed-variable drive 25. In this case, the higher the gear ratio of the toroidal infinite speed-variable drive 25, the higher the gear ratio of the entire infinite speed-variable drive.

During acceleration or constant-speed driving in such a high-speed mode, the torque having passed through the toroidal infinite speed-variable drive 25 is applied in the direction in which the input disks 2, 2 apply torque to the power rollers 20, 20.

[0018]

[Patent Document 1]

JP-A-2-283949

[Patent Document 2]

JP-A-8-4869

[Patent Document 3]

JP-A-11-303961

[Patent Document 4]

JP-A-2000-220719

[0019]

[Problems that the Invention is to Solve]

Patent Document 4 describing the infinite speed-variable drive such as that mentioned above discloses only the principle of the speed-variable drive but fails to disclose a specific structure. When the infinite speed-variable drive is embodied, a contrivance must be applied to a structure of a coupling section between one input disk 2 constituting the toroidal infinite speed-variable drive 25 and the carrier 27 constituting the planetary gear transmission 26. Specifically, in the case of an infinite speed-variable drive of a so-called geared neutral type, when the output shaft 38 is stopped or rotated at a very low speed while the input shaft 1 is being rotated, the torque passing through the toroidal infinite speed-variable drive 25 becomes extremely large. Accordingly, the coupling section must possess sufficient strength to transmit such large torque.

[0020]

However, as described in Patent Document 3, the structure in which the inner diameter portion of the outer surface of the output disk and the end section of the sleeve are coupled together through irregularity engagement is not necessarily capable of transmitting large torque, because the diameter of the torque transmission section is small. In contrast, Japanese

Patent Application No. 2001-246864 describes a structure, wherein torque can be transmitted between an input disk and a transmission member by means of causing a plurality of protuberances provided on a half section of the outer surface of the input disk, the protuberances projecting toward the outside rather than to the center section of the outer surface, to engage with an extremity section of a transmission projecting piece provided on a transmission member to be used for transmitting torque to the input disk. This structure of the prior invention is also intended for placing the protuberances at radially positions about 10 mm or thereabouts inside the outer edge of the input disk. Hence, large stress acts on portions of the input disk, possibly resulting in a failure to ensure sufficient durability.

An infinite speed-variable drive of the present invention has been conceived in light of the circumstances.

[0021]

[Means for Solving the Problem]

Accordingly, similar to the above-described known continuously variable transmission apparatus, the continuously variable transmission apparatus of the present invention, has an input shaft which is connected to and is rotatively driven by a drive source, an output shaft for extracting power based on rotation of the input shaft, and a toroidal infinite speed-variable transmission and a planetary gear transmission.

The toroidal infinite speed-variable transmission comprises a pair of input disks, each having inner and outer side surfaces and rotating in association with rotation of the input shaft; an output disk which has a side surface, which is disposed concentrically with the respective input disks around an intermediate section of the input shaft while the side surface is oriented opposite the inner side surfaces of the respective input disks, and which is rotatable independently of the input disks; plural trunnions which are associated with each pair of said input disk and said output disk, each trunnion interposed between the input disk and the output disk to sway around an axle located at a position offset with respect to the center axles of the disks; support shafts each associated with each of the trunnions and protruding from an inner side surface of the trunnion; a power roller each sandwiched between the input disk and the output disk while being supported rotatively by a support shaft and each being associated with each of the trunnions.

The planetary gear mechanism is composed by engaging a planetary gear with a sun gear and a ring gear, the planetary gear disposed between the sun gear and the ring gear which is disposed around the sun gear, the planetary gear rotatively supported by said carrier which is supported concentrically with said sun gear and supported on one end section of said input shaft.

[0022]

Especially, in the infinite speed-variable drive according to the present invention, transmission of power between said carrier and said one input disk is made free by means of causing a plurality of projection sections to engage with a plurality of engagement sections,

The plurality of projection sections are projectingly provided at positions closer to the radially outside than to a circumscribed circle of a contact oval existing in said traction section.

[0023]

[Operation]

The infinite speed-variable drive of the present invention having the foregoing construction transmits power between the input shaft and the output shaft. The basic operation required when a gear ratio of the input shaft to the output shaft is changed is the same as that adopted in the case of the conventionally-known infinite speed-variable drive shown in Fig. 9.

Particularly, in the case of the infinite speed-variable drive of the present invention, a plurality of protuberances projectingly provided on a portion of an outer surface of one input disk are engaged with a plurality of engagement sections formed on a part of the carrier. Large torque can be transmitted between the carrier and the one disk.

However, the respective protuberances are provided at positions closer to the outside diameter than to the diameter of a pitch circle of a traction section. The stress applied to one input disk during transmission of torque is suppressed to a low level, thereby sufficiently ensuring durability of respective constituent members including the input disk.

[0024]

[Mode for Implementing the Invention]

Figs. 1 through 7 show an embodiment of the present invention. In the drawings, dimensional relationships, such as aspect ratios, are shown in an actual scale. An infinite speed-variable drive of the present embodiment is constituted by combination of a toroidal infinite speed-variable transmission 25a and first through third planetary gear transmissions 42 to 44 and has an input shaft 1a and an output shaft 38a. Of these elements, first and second planetary gear transmissions 42, 43 correspond to planetary gear mechanisms described in claims. In the illustrated embodiment, a transmission shaft 45 is provided concentrically with and between the input shaft 1a and the output shaft 38a such that the transmission shaft 45 is rotatable relative to the shafts 1a and 38a. The first and second planetary gear transmissions 42, 43 are provided so as to bridge the input shaft 1a and the transmission shaft 45, and the third planetary gear transmission 44 is provided so as to bridge the transmission shaft 45 and

the output shaft 38a.

[0025]

Of these elements, the toroidal infinite speed-variable transmission 25a comprises a pair of input disks 2a, 2b; an integrated output disk 17b; and a plurality of power rollers 20, 20. The pair of input disks 2a, 2b are coupled concentrically with each other by way of the input shaft 1a so as to be able to rotate synchronously. The output disk 17b is interposed between the input disks 2a, 2b concentrically therewith and supported such that the output disk 17b is rotatable relative to the input disks 2a, 2b. Moreover, the plurality of power rollers 20, 20 are sandwiched between the axial side surfaces of the output disk 17b and single-axial-side surfaces of the input disks 2a, 2b. In association with rotation of the input disks 2a, 2b, power is transmitted from the input disks 2a, 2b to the output disk 17b.

[0026]

In the case of the present embodiment, both axial end sections of the output disk 17b are rotatively supported by roll bearings, such as a pair of thrust angular ball bearings 46, 46. Therefore, in the case of the present embodiment, a pair of supports 48, 48 are provided in a casing 11 by way of an actuator body 47 in order to support a pair of support plates 24, 24 intended for supporting both end sections of the respective power rollers 20, 20. Each of the supports 48, 48

is constituted by coupling together a pair of support post sections 49a, 49b, which are provided on both sides with reference to the input shaft 1a in a concentric manner, by means of an annular support ring section 50. The input shaft 1a is inserted into the support ring section 50.

[0027]

Lower end sections of the respective supports 48, 48 are fixedly coupled to an upper surface of the actuator body 47 with a plurality of bolts 51, 51 while limitations are imposed on the position and direction in which the supports are to be mounted. Therefore, recessed sections 52, 52 to be used for receiving the lower end sections of the respective supports 48, 48 without rattle are formed in the upper surface of the actuator body 47. A plurality of screw holes are opened in the lower end faces of the respective lower end sections of the supports 48, 48. The respective supports 48, 48 are fixed at predetermined positions on the upper surface of the actuator body 47, by means of inserting the lower end sections of the supports 48, 48 into the actuator body 47 from below and screw-engaging the lower end sections with the respective recessed sections 52, 52 while the lower end sections of the supports 48, 48 are fitted into the inside of the respective recessed sections 52, 52, and fastening the lower end sections through use of the bolts 51, 51.

[0028]

Upper end sections of the respective supports 48, 48 are fixedly coupled to a lower surface of a coupling plate 53 by means of bolts 54, 54 while limitations are imposed on the mount positions of the upper end sections. Therefore, recessed sections 55, 55 to be used for receiving the upper end sections of the respective supports 48, 48 without rattle are formed in the lower surface of the coupling plate 53. One thread hole is formed in the center of the upper end surface of each of the support sections 48, 48. The respective supports 48, 48 are fixed at predetermined positions on the lower surface of the coupling plate 53, by means of inserting the upper end sections of the supports 48, 48 into the coupling plate 53 from above and screw-engaging the upper end sections with the respective thread holes while the upper ends of the supports 48, 48 are fitted into the inside of the respective recessed sections 55, 55, and fastening the upper end sections through use of the bolts 54, 54.

[0029]

As mentioned above, the pair of supports 48, 48 are fixedly coupled between the upper surface of the actuator body 47 and the lower surface of the coupling plate 53 so as to bridge them while imposing limitations on the positions of the supports. In this state, of the support post sections 49a, 49b provided in the vicinity of respective end sections of the respective supports 48, 48, the lower support post sections 49a, 49a are

located at positions immediately above the upper surface of the actuator body 47. Support holes 56a, 56a formed in the lower support plate 24 of the pair of support plates 24, 24 are fitted around the support post sections 49a, 49a of the supports 48, 48 without rattle. The upper support post sections 49b, 49b are located at positions immediately below the lower surface of the coupling plate 53. Support holes 56b, 56b formed in the upper support plate 24 of the pair of support plates 24, 24 are fitted around support post sections 49b, 49b of the supports 48, 48 without rattle. The power rollers 20, 20 are rotatively supported between the support plates 24, 24, which have been provided in the aforementioned manner, by way of a plurality of trunnions 22, 22 and support shafts 23, 23. A circumferential surface 21 of each power roller 20 is brought into rolling contact with an input-side surface 3 of the input disk 2 and an output side surface of the output disk 17.

[0030]

Of the actuator body 47 and the coupling plate 53 coupled together by means of the pair of supports 48, 48, the actuator body 47 is fixedly supported below the casing 11 while limitations are imposed on the lengthwise and widthwise positions of the actuator body 47 (i.e., the horizontal direction in Figs. 1 and 2 and the direction from the front to rear side of Fig. 3), and the coupling plate 53 is fixedly supported within the casing 11 while limitations are imposed on the lengthwise

and widthwise positions of the coupling plate 53 (i.e., the horizontal direction in Figs. 1 and 2 and the direction from the front to rear side of Fig. 3). In the illustrated embodiment, cylindrical positioning sleeves 59, 59 are provided so as to bridge positioning recessed sections 58a, 58b formed in portions of the upper surface of the coupling plate 53 and portions of the lower surface of a ceiling section 57 of the casing 11, the portions opposing each other. The coupling plate 53 is positioned through use of a plurality of unillustrated positioning pins.

[0031]

In this way, the support ring sections 50, 50 are present in the centers of the respective cavities (spaces) which are located between the pair of supports 48, 48 fixed to predetermined positions within the casing 11 and defined between the input disks 2a, 2b and the output disk 17b. By means of the support ring sections 50, 50, the output disk 17b is rotatively supported. Therefore, the thrust angular ball bearings 46, 46 are provided between axial end faces of the respective support rings 50, 50 and both axial end surfaces of the output disk 17b; that is, locations closer to the inner diameter than to the output side surfaces 18, 18 provided on both axial side surfaces of the output disk 17b.

[0032]

In the case of the unillustrated infinite speed-variable

transmission, a base end section (i.e., the left end section in Fig. 1) of the input shaft 1a is coupled to a crank shaft of an unillustrated engine by way of a drive shaft 60. By means of the crank shaft, the input shaft 1a is rotatively driven. A hydraulic pressing apparatus is used as a pressing apparatus 6a to be used for applying appropriate surface pressure to roll contact sections (traction sections) between the input side surfaces 3, 3 of the input disks 2a, 2b, the output side surfaces 18, 18 of the output disk 17b, and the circumferential surfaces 21, 21 of the power rollers 20, 20. Pressure oil is freely supplied to the pressing apparatus 6a and hydraulic actuators 79, 79 to be used for displacing the power rollers 20, 20 for changing speed, from a pressure source of a gear pump or the like. Pressure oil is freely supplied to a hydraulic cylinder in order to disconnect a low-speed clutch 61 and a high-speed clutch 62, which will be described later, from a pressure source of a gear pump or the like.

[0033]

A base end section (i.e., the left end section in Figs. 1 and 2) of a hollow rotary shaft 32a is coupled to the output disk 17 by means of spline engagement. The hollow rotary shaft 32a is inserted into the (one) input side disk 2b that is located at a position distal from the engine (i.e., the right side in Figs. 1 and 2), thereby enabling extraction of rotational force of the output disk 17b. Moreover, a first sun gear 63 intended

to be used for constituting the first planetary gear transmission 42 is fixedly provided at a portion of the extremity (i.e., the right side portion in Figs. 1 and 2) of the hollow rotary shaft 32a which protrudes from the outside surface of the input disk 2.

[0034]

A first carrier 64 as a carrier described in claims is provided so as to bridge the projecting portion of the hollow rotary shaft 32a and the input disk 2b at the extremity section (i.e., the right end in Figs. 1 and 2) of the input shaft 1a, thereby causing the input disk 2b to rotate synchronous with the input shaft 1a. The first carrier 64 has three support plates 65a to 65c which assume the shape of a disk and are axially spaced from and arranged concentrically with each other. A cylindrical support cylinder section 66 is fixedly provided on an inner periphery of the support section 65b arranged at an axially intermediate position in the three support plates 65a to 65c. The support cylinder section 66 is coupled to the extremity section of the input shaft 1a through spline engagement. Further, an extremity edge (i.e., the right edge in Fig. 1) of the support cylinder section 66 is suppressed by a nut 67, to thus fix the support cylinder section 66 to the input shaft 1a.

[0035]

In contrast, the support plates 65a, 65c, which are located

on the respective ends in the axial direction, are fixed to the intermediate support plate 65b by means of planetary shafts 68a, 68b and directly coupled together by means of a planetary shaft 68c. Specifically, one-end sections of the planetary shafts 68a (i.e., the right end section in Figs. 1 and 4) are fixedly fitted to positions (in general, three to four positions) spaced at uniform intervals on one side surface of the intermediate support plate 65b opposing the input disk 2b. The support plate 65a (i.e., the left side in Figs. 1, 2, and 4) is fixedly fitted to the other end sections (i.e., the left end sections in Figs. 1 and 4) of the respective planetary shafts 68a. One-end sections of the planetary shafts 68b (i.e., the left end section in Figs. 1 and 4) are fixedly fitted to positions (in general, three to four positions) spaced at uniform intervals on one side surface of the intermediate support plate 65b opposing the input disk 2a. The other support plate 65c (the right side in Fig. 1) is fixedly fitted to the other end sections of the respective planetary shafts 68b (i.e., the right end sections in Fig. 1). Both end sections of the remaining planetary shaft 68c are fixedly fitted to positions closer to the outside diameter than to the positions of the support plates 65c and 64b (i.e., three to four locations in general) which are circumferentially spaced at uniform intervals. A portion of the intermediate support section 65b aligned with the remaining axle 68c and the surroundings of that portion are

cut away. In order to transmit large torque and to enhance rigidity and strength of the first carrier 64, the support plates 65a to 65c are coupled together by means of the respective planetary shafts 68a to 68c and additional support sections.

[0036]

Notches 69 formed in a plurality of areas (three areas in the embodiment) on the outer peripheral edge of the support plate 65a, the areas being spaced at uniform intervals, and a plurality of protuberance sections 70, 70 (three protuberance sections in the embodiment) are engaged with each other. As shown in Figs. 5 and 6, the protuberance sections 70, 70 are formed at positions on the outer side surface of the input disk 2b (i.e., the right side surface in Figs. 1, 2, 4, 6, and 7), the positions being circumferentially spaced at uniform intervals. More specifically, as shown in Fig. 7A, a roll contact section between the input side surface 3 serving as an inner surface of the input disk 2b and the circumferential surface 21 of the power roller 20 is a traction section. When the traction section is located at a position on the input side surface 3 close to the outside diameter (i.e., in the state of maximum acceleration of the toroidal infinite speed-variable transmission 25a), the projection sections 70, 70 are provided projectingly at positions closer to the outside diameter than to a diameter Dp of a pitch circle of the traction section. The protuberance sections 70, 70 are engaged with the notches

69 formed in the support plate 65a without rattle in at least the circumferential direction. As a result, power is freely transmitted between the first carrier 64 and the input disk 2b.

[0037]

Planetary gears 71a to 71c are rotatively supported on the respective planetary shafts 68a to 68c provided on the first carrier 64, thereby constituting the first and second planetary gear transmissions 42, 43 of double pinion type. Moreover, a first ring gear 72 is rotatively supported around half the area of the first carrier 64 (i.e., a right half of the first carrier in Figs. 1 and 2). Of the planetary gears 71a to 71c, the planetary gear 71a—which is disposed close to the toroidal infinite speed-variable transmission 25a (i.e., the left-side area in Figs. 1 and 2) and inside in the radial direction of the first carrier 64—is engaged with the first sun gear 63. The planetary gear 71b—which is disposed at a position distal from the infinite speed-variable transmission 25a (i.e., the right side portion in Figs. 1 and 2) and inside in the radial direction of the first carrier 64—is engaged wit the second sun gear 73 fixed on the base end section (a left end section in Fig. 1) of the transmission shaft 45. The remaining planetary gear 71c disposed outside in the radial direction of the first carrier 64 is made larger in axial dimension than the planetary gears 71a, 71b disposed inside and is engaged with the planetary

gears 71a, 71b. Moreover, the remaining planetary gear 71c is meshed with the first ring gear 72. Instead of causing the planetary gears located close to the outside in the radial direction to be independent of each other within the first and second planetary gear transmissions 42, 43, there may also be employed a structure in which a ring gear having a larger width is meshed with the planetary gears.

[0038]

A second carrier 74 to be used for constituting the third planetary gear transmission 44 is fixedly coupled to the base end section (i.e., the left end section in Fig. 1) of the output shaft 38a. The second carrier 74 and the first ring gear 72 are coupled together by way of the low-speed clutch 61. A third sun gear 75 is fixedly provided at an extremity portion of the transmission shaft 45 (i.e., the right end portion in Figs. 1 and 2). A second ring gear 76 is disposed around the third sun gear 75. The high-speed clutch 62 is provided at an area where the second ring gear 76 and the casing 11 or the like are fixed together. Moreover, a plurality of sets of planetary gears 77a, 77b interposed between the second ring gear 76 and the third sun gear 75 are rotatively supported by the second carrier 74. These planetary gears 77a, 77b mesh with each other, and the planetary gear 77a disposed inside in the radial direction of the second carrier 74 is engaged with the third sun gear 75, and the planetary gear 77b disposed outside is

engaged with the second ring gear 76.

[0039]

In the case of the infinite speed-variable drive of the present embodiment having the foregoing construction, power is transmitted from the input shaft 1a to the integral output disk 17b by way of the pair of input disks 2a, 2b and the power rollers 20, 20. The thus-transmitted power is extracted by way of the hollow rotary shaft 32a. In the low-speed mode in which the low-speed clutch 61 is connected and the high-speed clutch 62 is disconnected, the rotational speed of the output shaft 38a can be switched between forward rotation and rearward rotation with a stop therebetween while the rotational speed of the input shaft 1a is kept constant, by means of changing the gear ratio of the toroidal infinite speed-variable transmission 25a. Specifically, in this low-speed mode, a difference between the first carrier 64 rotating forward along with the input shaft 1a and the first sun gear 63 rotating in reverse along with the hollow rotary shaft 32a is transmitted from the first ring gear 72 to the output shaft 38a by way of the low-speed clutch 61 and the second carrier 74. In this state, the output shaft 38a is stopped by means of changing the gear ratio of the toroidal infinite speed-variable transmission 25a to a predetermined value. In addition, the output shaft 38a is also rotated in a direction in which the automobile is to run rearward, by means of changing the gear

ratio of the toroidal infinite speed-variable transmission 25a toward acceleration from the predetermined value. In contrast, the output shaft 38a is rotated in a direction in which the automobile is to run forward, by means of changing the gear ratio of the toroidal infinite speed-variable transmission 25a toward deceleration from the predetermined value.

[0040]

In the high-speed mode in which the low-speed clutch 61 is disconnected and the high-speed clutch 62 is connected, the output shaft 38a is rotated in a direction in which the automobile is to run forward. Specifically, in this high-speed mode, rotation of the planetary gear 71a of the first planetary gear transmission 42, the planetary gear rotating in accordance with a difference in rotational speed between the first carrier 64—which rotates forward along with the input shaft 1a—and the first sun gear 63—which rotates reversely along with the hollow rotary shaft 32a and the first carrier 64—is transmitted to the planetary gear 71 of the second planetary gear transmission 43 by way of another planetary gear 71c. As a result, the transmission shaft 45 is rotated by way of the second sun gear 73. The second carrier 74 and the output shaft 38a coupled with the second carrier 74 rotate forward by means of engagement between the third sun gear 75 provided at the extremity of the transmission shaft 45, the second ring gear 76 constituting the third planetary gear transmission 44 along

with the third sun gear 75, and the planetary gears 77a, 77b. In this state, as the gear ratio of the toroidal infinite speed-variable transmission 25a is changed toward acceleration, the rotational speed of the output shaft 38a can be increased.

[0041]

In the case of the infinite speed-variable drive of the present embodiment, transmission of power between the input disk 2b and the first carrier 64 can be performed without fail while an attempt is realized to achieve a reduction in size and weight. Specifically, in order to perform power transmission, the plurality of protuberance sections 70, 70 projectingly provided on the outer side surface of the input disk 2b are formed along an outer edge of the outer side surface at positions closer to the outside in the radial direction than to the pitch diameter Dp of the traction section. The notches 69 formed in the outer edge of the support plate 65a constituting the carrier 64 are engaged with the protuberance sections 70, 70. Portions constituting the notches 69 are provided at positions more toward the outside in the radial direction than to the areas where the edge sections of the respective planetary shafts 68a are fitted. Therefore, as shown in Fig. 4A, interference between the protuberance sections 70, 70 and the end sections of the respective planetary shafts 68a does not arise even when the thickness of the support plate 65a is not made larger than necessary to ensure strength.

[0042]

For this reason, an attempt can be realized to make the transmission compact and lightweight by means of rendering the support plate 65a thin. In the case of a structure relating to Japanese application No. 2001-246864, the protuberance sections 70a are placed at positions radially inside about 10 mm or thereabouts from the outer edge of the input disk 2b, as shown in Fig. 4B. In order to prevent occurrence of interference between the protuberance sections 70a and the planetary shafts 68a, there arise a necessity for increasing the thickness of a support plate 65a′ and forming, in one surface of the support plate 65a′, recessed sections 78 to be used for engaging the protuberance sections 70a. For this reason, in the case of the foregoing structure, difficulty is encountered in making an attempt to achieve a reduction in size and weight by making the support plate 65a′ thin. Machining operations required to make the recessed sections 78 are more troublesome than the operations for producing the notches, thus adding to costs. According to the structure of the present invention, the diameter of the portion of the transmission to be used for transmitting torque (i.e., the engagement section between the protuberance sections 70, 70 and the notches 69) is large. Hence, when torque of same magnitude is transmitted, the force acting on the engagement section can be suppressed to a level of force smaller than that required in the foregoing prior invention.

Even in this regard, the structure of the present invention is advantageous over the previously-described prior invention.

[0043]

Moreover, in the case of the present invention, the protuberance sections 70, 70 are provided at positions radially outside the pitch circle diameter Dp of the traction section, and hence the durability of the input disk 2b can also be improved. The reason for this is that a displacement arises between a portion of the input disk 2b where stress is exerted in association with transmission of torque between the input disk 2b and the first carrier 64 and a portion of the input disk 2b where stress is exerted in association with transmission of torque between the input disk 2b and the power rollers 20. Specifically, shearing stress acts on the portions of the input disk 2b where the protuberance sections 70, 70 are formed in association with transmission of torque between the input disk 2b and the first carrier 64. Further, when torque is transmitted between the power rollers 20 and the input disk 2b, compression and shearing stress act on the traction section. In the case of the present invention, the radial positions of the protuberance sections 70, 70 are limited in the manner mentioned previously. Hence, as shown in Fig. 7A, in association with transmission of two types of torque, the stress acts on different locations on the input disk 2b (i.e., not an overlapping location). Therefore, an attempt can be easily realized to

improve fatigue life of metal constituting the input disk 2b. In contrast, in the case of the above prior invention, the stress acts on an overlapping position on the input disk 2b in association with transmission of two types of torque, as shown in Fig. 7B. For this reason, difficulty is encountered in realizing an attempt to improve the fatigue life of the metal constituting the input disk 2b. Even in this aspect, the structure of the present invention is advantageous over the foregoing prior invention.

[0044]

When the input side surface 3, which is an inner surface of the input disk 2, is finished, an unillustrated backup plate is abutted against the inner-diameter portions of the respective protuberance sections 70, 70 on the outer side surface of the input disk 2b. A diameter D70 of the inner-diameter portions (i.e., inscribing circles of the respective protuberance sections 70, 70) of the protuberance sections 70, 70 is larger than the pitch circle diameter Dp of the traction section (D70 > Dp). Hence, the support strength of the portion of the input side surface 3 which can come into contact with the circumferential surface 21 of the power roller 20 can be sufficiently ensured. For this reason, even in view of ensuring the shape and dimensional accuracy of the input side surface 3, the structure of the invention is advantageous over the previously-described prior invention. Particularly, if the

respective protuberance sections 70, 70 are provided at positions radially outside the circumscribing circle of the contact ellipse existing in the traction section, the structure of the invention becomes advantageous in terms of ensuring precision of the input side surface 3 and ensuring fatigue life.

[0045]

[Advantage of the Invention]

The present invention is configured and operates in the manner as mentioned previously. Hence, there can be realized an infinite speed-variable drive capable of transmitting large power, having superior durability, and being compact and lightweight.

[Brief Description of the Drawings]

[Fig.2]

Fig. 1 is a cross-sectional view showing one example of an embodiment of the present invention;

[Fig.2]

Fig. 2 is an enlarged view of the center of Fig. 1;

[Fig.3]

Fig. 3 is an enlarged cross-sectional view taken along line A-A shown in Fig. 1;

[Fig.4]

Fig. 4A is an enlarged view of the structure of the infinite speed-variable drive of the present invention shown in Fig. 1B; Fig. 4B is an enlarged view of the infinite speed-variable

drive to which the prior invention is applied shown in Fig. 1B;

[Fig.5]

Fig. 5 is a view of an input disk when viewed in the right in Fig. 4;

[Fig.6]

Fig. 6 is a cross-sectional view taken along line C-C shown in Fig. 5;

[Fig.7]

Fig. 7A is a partially-omitted cross-sectional view of the structure of the infinite speed-variable drive of the present invention along line D-D shown in Fig. 1; Fig. 7B is a partially-omitted cross-sectional view of the infinite speed-variable drive to which the prior invention is applied shown in Fig. 1B;

[Fig.8]

Fig. 8 is a side view showing an example of a conventionally-known toroidal infinite speed-variable transmission when it is in the state of maximum deceleration; and

[Fig.9]

Fig. 9 is an essentially-cross-sectional view showing an example of a conventionally-known infinite speed-variable drive.

[Descriptions of the Reference Numerals]

1, 1a INPUT SHAFTS

2, 2a, 2b INPUT DISKS

3 INPUT SIDE

4 BALL SPLINE

5 ROLL BEARING

6, 6a PRESSING APPARATUS

7 CAM PLATE

8 DRIVE SHAFT

9 LOADING NUT

10 FLAT LEAF SPRING

11 CASING

12 PARTITION SECTION

13 THROUGH HOLE

14 OUTPUT CYLINDER

15 ROLL BEARING

16 OUTPUT GEAR

17, 17a, 17b OUTPUT DISKS

18 OUTPUT SIDE SURFACE

19 NEEDLE BEARING

20 POWER ROLLER

21 CIRCUMFERENTIAL SURFACE

22 TRUNNION

23 SUPPORT SHAFT

24 SUPPORT PLATE

25, 25a TORODIAL INFINITE SPEED-VARIABLE TRANSMISSION

26 PLANETARY GEAR TRANSMISSION

27 CARRIER

28a, 28b PLANETARY GEARS

29 FIRST TRANSMISSION SHAFT

30a, 30b SUN GEARS

31 SECOND TRANSMISSION SHAFT

32, 32a HOLLOW ROTARY SHAFTS

33 SUN GEAR

34 PLANETARY GEAR

35 RING GEAR

36 SECOND CARRIER

37a, 37b OUTPUT SHAFTS

39 SECOND RING GEAR

40 LOW-SPEED CLUTCH

41 HIGH-SPEED CLUTCH

42 FIRST PLANETARY GEAR TRANSMISSION

43 SECOND PLANETARY GEAR TRANSMISSION

44 THIRD PLANETARY GEAR TRANSMISSION

45 TRANSMISSION SHAFT

46 THRUST ANGULAR BALL BEARING

47 ACTUATOR BODY

48 SUPPORT

49a, 49b SUPPORT POST SECTIONS

50 SUPPORT RING SECTION

51 BOLT

52 RECESSED SECTION

53 COUPLING PLATE

54 BOLT

55 RECESSED SECTION

56a, 56b SUPPORT HOLES

57 CEICLING PLATE

58a, 58b POSITIONING RECESS

59 POSITIONING SLEEVE

60 DRIVE SHAFT

61 LOW-SPEED CLUTCH

62 HIGH-SPEED CLUTCH

63 FIRST SUN GEAR

64 FIRST CARRIER

65a, 65b, 65c, 65a' SUPPORT PLATE

66 SUPPORT CYLINDER SECTION

67 NUT

68a, 68b, 68c PLANETARY SHAFTS

70, 70a PROTUBERANCE SECTIONS

71a, 71b, 71c PLANETARY GEARS

72 FIRST RING GEAR

73 SECOND SUN GEAR

74 SECOND CARRIER

75 THIRD SUN GEAR

76 SECOND RING GEAR

77a, 77b PLANETARY GEARS

78 RECESSED SECTION

79 ACTUATOR

[Designation of Document] Abstract

[Abstract]

[Problem] To improve the torque transmission mechanism between the input disk 2b constituting a troidal continuously variable transmission apparatus and the first carrier 64 constituting the planetary gear mechanism not only to miniaturize/reduce the weight of the apparatus but to secure a durability of said input disk 2b.

[Means for Solution] A plurality of protuberances 70 are formed on an outer surface of this input disk 2b, which is an outer edge portion that is located at a radially outer side of a traction portion that is positioned on a side of the inner surface. Each of the plurality of protuberances 70 is engaged with the notch formed on an outer edge portion of the support plate 65a of the first carrier 64. By this structure, it is possible to reduce the stress acted on the input disk 2b and make the support plate 65a thinner, so that the above problem is solved.

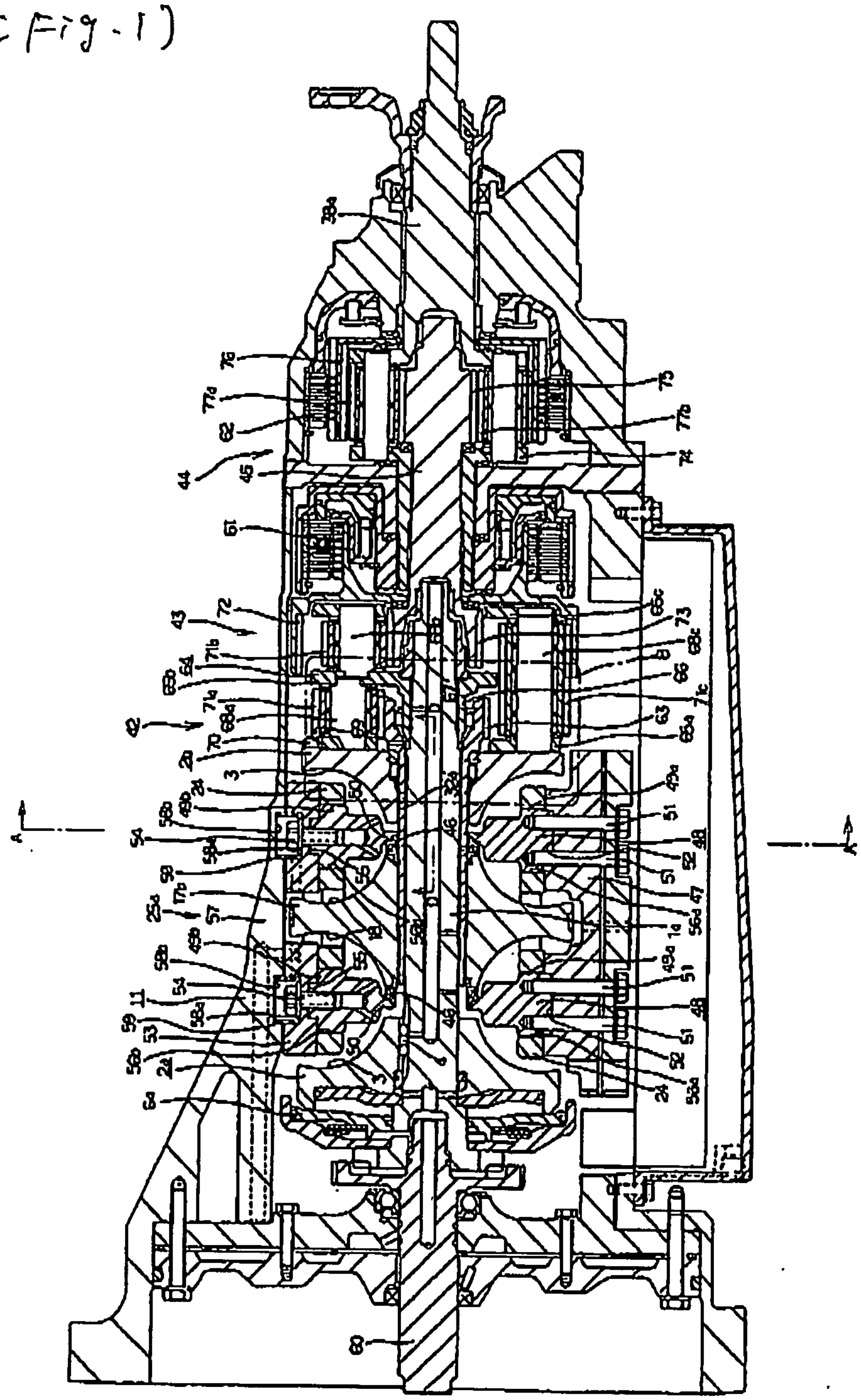
[Selected Drawing] Fig. 4

[Document Name] Drawings

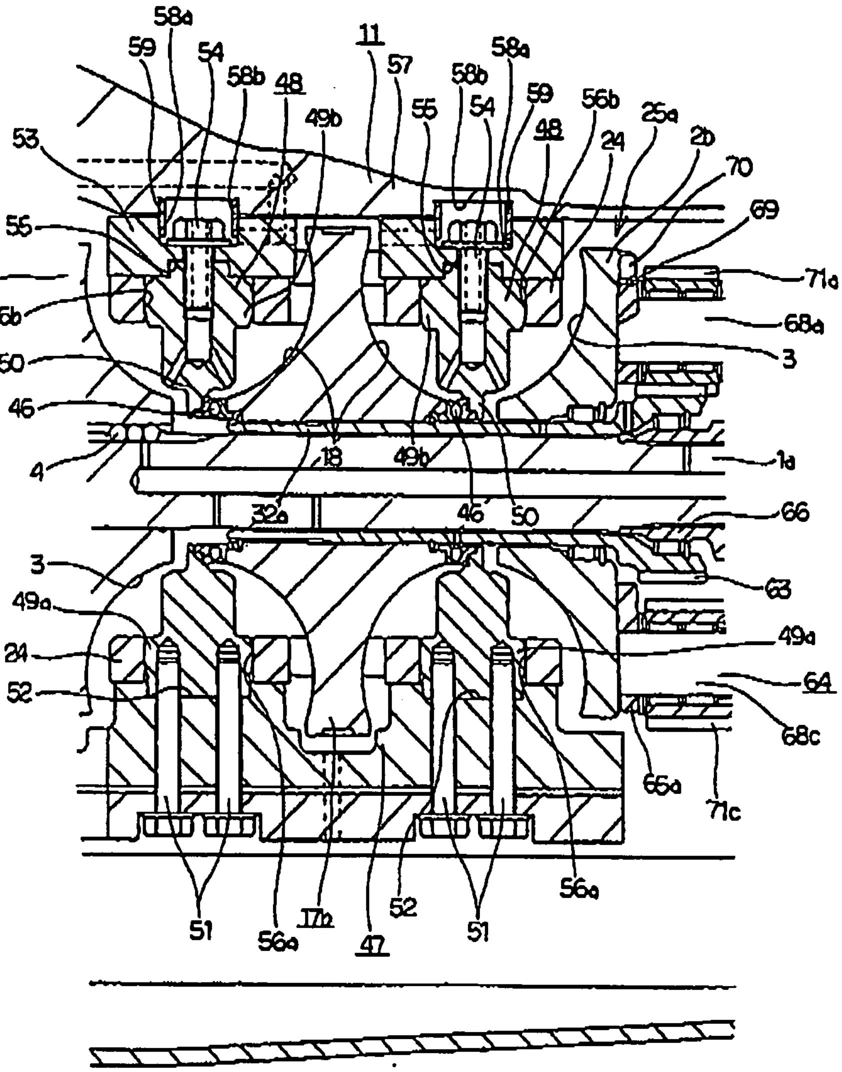
~~【書類名】 図面~~

~~【図1】~~

[Fig. 1]



【図２】
[Fig-2)



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50
3
46
4
18
1a
32a
66
63
49a
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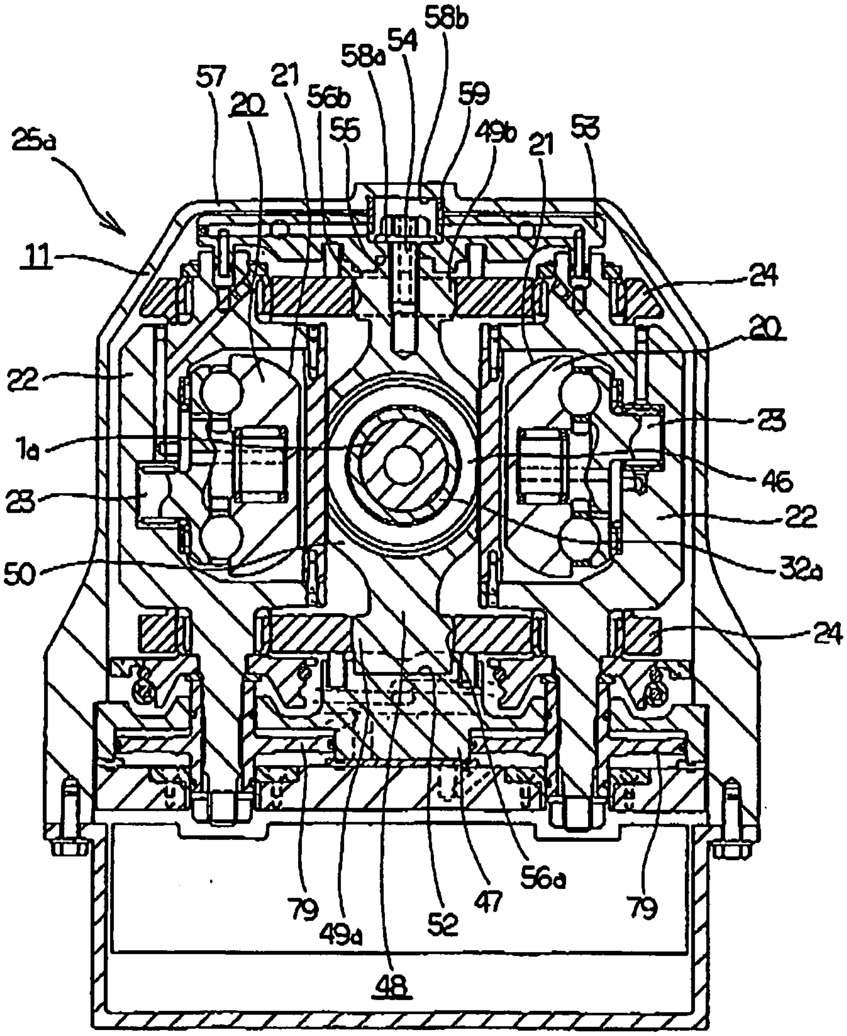
52
64
68c
65a
71c
51
56a
17b
47

【図３】

CFig-3)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49a
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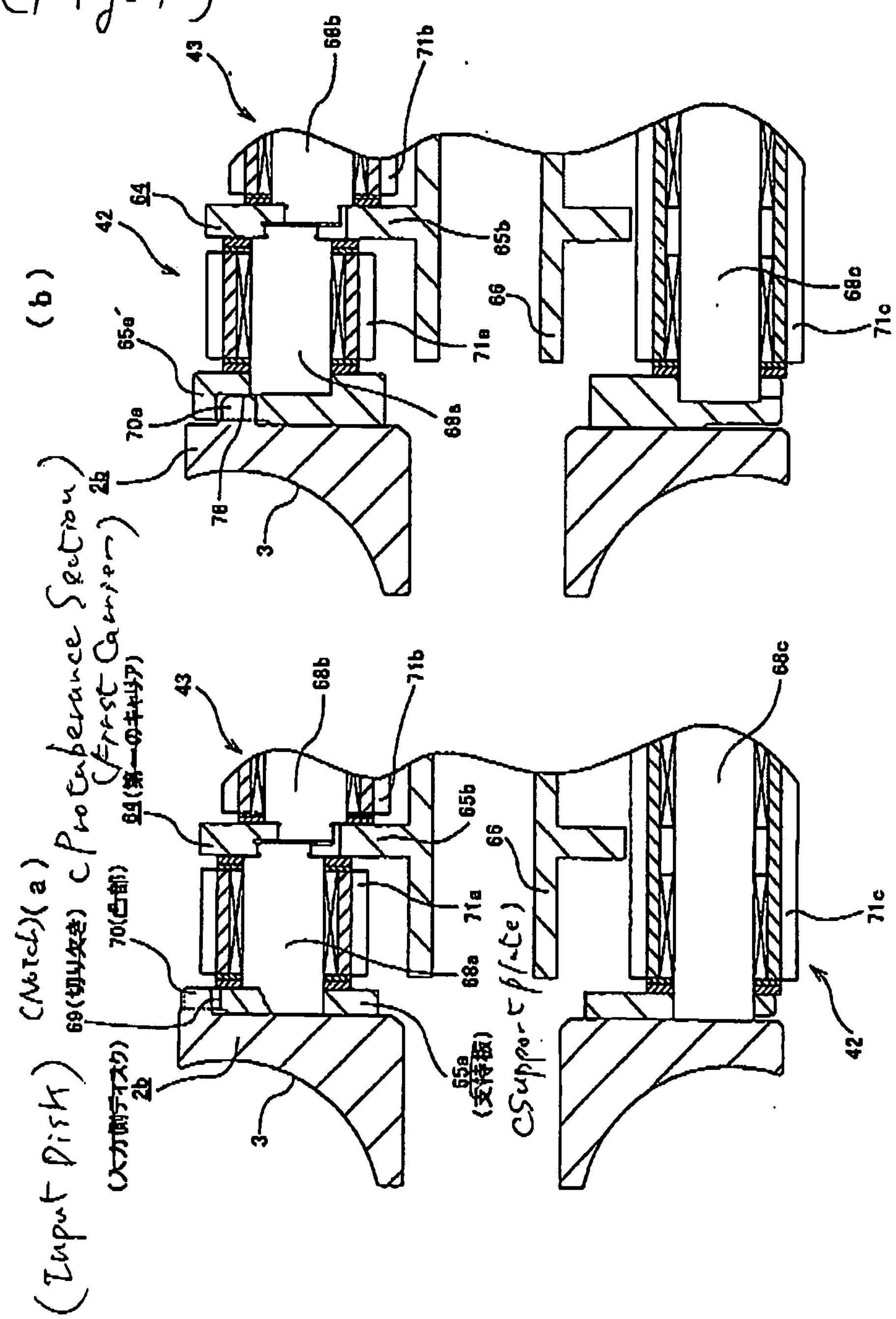
【図4】
(Fig. 4)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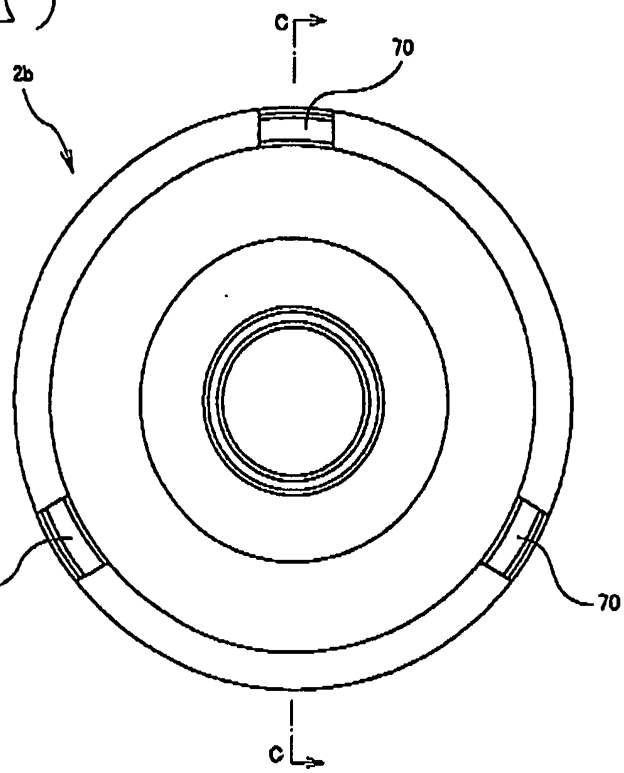
68b
71b
64
42
65b
66
68c
71c
71a
65a'
70a
68a
2b
76
3
(a)
64(第一のキャリア)
(First Carrier)
(Protuberance Section)
70(凸部)
(Notch)
69(切り欠き)
2b(入力側ディスク)
(Input Disk)
65a(支持板)
(Support Plate)

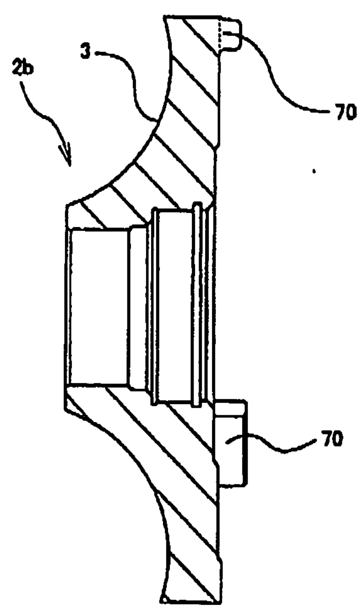
【図５】

(Fig.5)



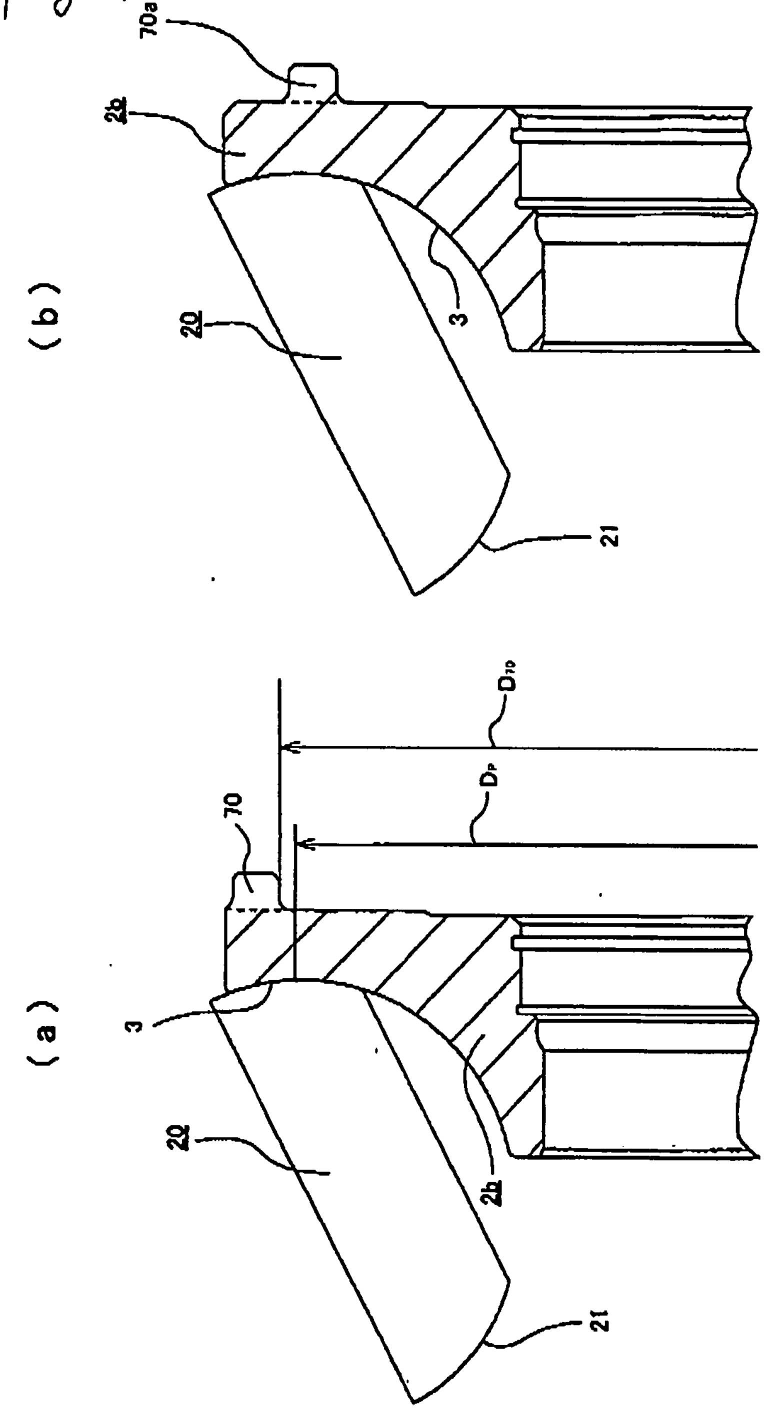
【図６】

(Fig.6)



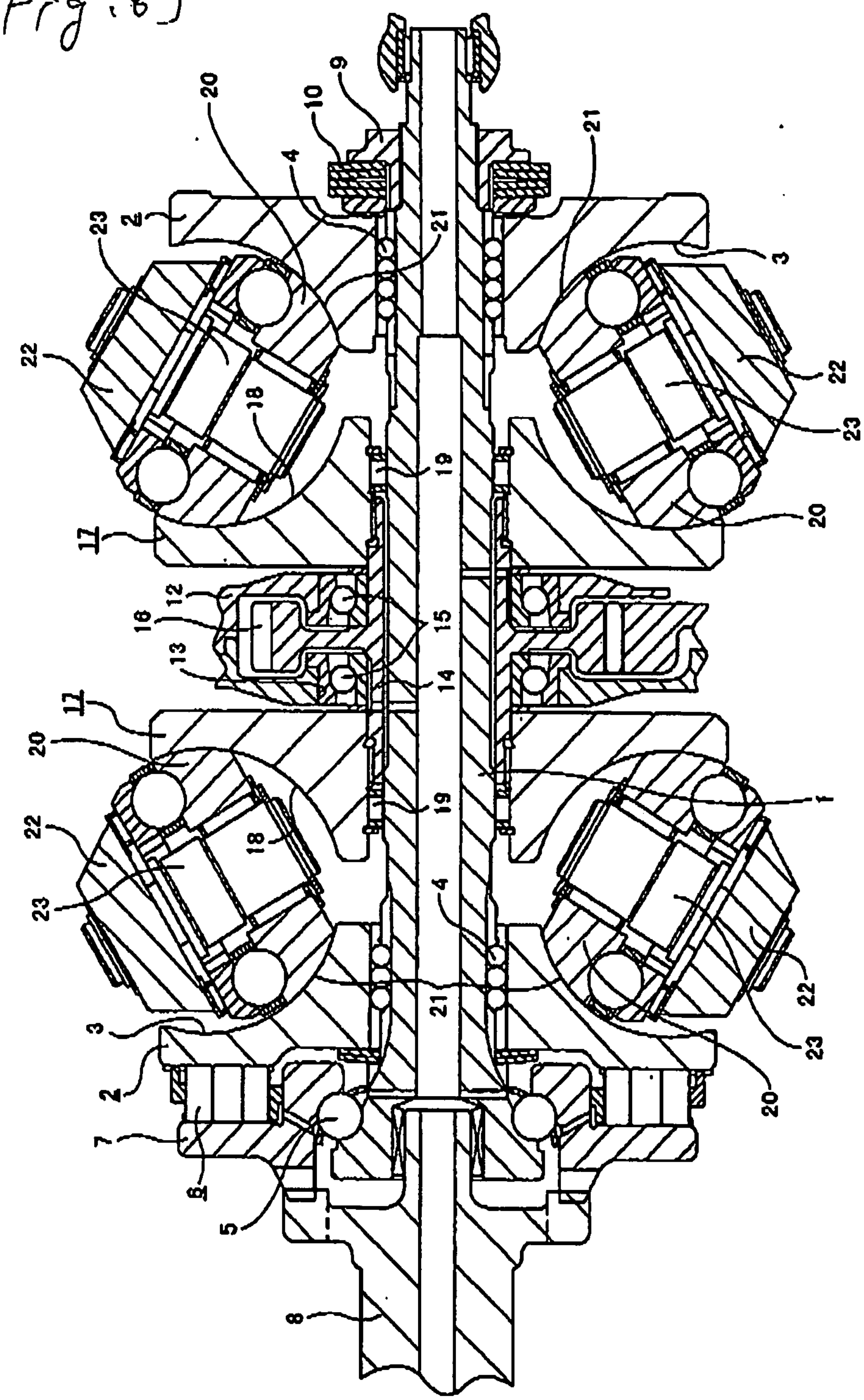
【図7】

CFig.7]



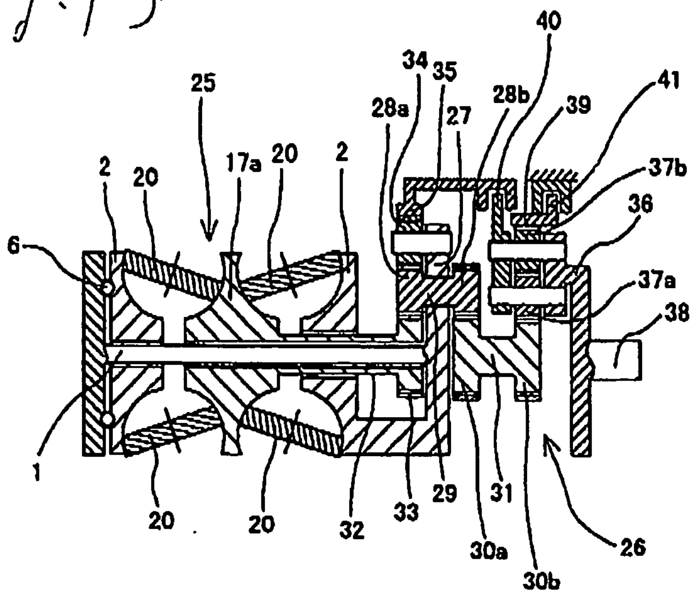
【図８】

(Fig.8)



【図９】

(Fig. 9)



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ARCHIVAL RECORD OF THE APPLICANT

Identification No. 000004204

1. Date of Change August 29, 1990

[Reason for Change] New Registration

Residence 6-3, Osaki 1-Chome, Shinagawa-ku, Tokyo-to.

Name NSK Ltd.